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Principles of Refrigerant Circuit Optimization in Single Row Microchannel Condensers

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ABSTRACT

Automotive HVAC systems manufacturers were the first to introduce microchannel condensers in the comfort air-conditioning market. HVAC&R original equipment manufacturers have followed their lead and recently started using microchannel condensers in residential and commercial cooling and heat pump applications. These applications began less than a decade back and their number has been increasing steadily (Mehendale, 2013). The main difference between microchannel condensers used for the automotive and residential / commercial applications is that the latter span a much wider range of coil face areas compared to the former. Considering this fact, there are relatively few well-documented studies in the public domain on the numerical modeling of two-phase heat transfer and pressure drop in microchannel condensers, especially in the size range of interest for residential / commercial climate control applications. More specifically, the principles behind the selection of the correct refrigerant circuit patterns that lead to the optimal thermal-hydraulic performance of microchannel condensers with different sizes is a matter that has not been satisfactorily addressed.

In this work, CoilDesigner™, an element-by-element heat exchanger model, was employed for simulating a single-row microchannel refrigerant-to-air condenser consisting of microchannel tubes and multi-louvered fins. Starting with a given microchannel tube design and a fixed multi-louvered fin geometry, the model was exercised to simulate the thermal-hydraulic performance of various two-, three-, and four-pass circuit configurations over a range of coil face areas. Based on the tradeoffs encountered between condenser heat duty and refrigerant pressure drop or pumping power required, recommendations have been provided to help designers and researchers in the HVAC&R field to select the best possible circuiting arrangement.

1. INTRODUCTION

Microchannel coils, especially condenser coils, have been in use in the automotive industry for more than twenty years. However, the technology has been adapted and applied to residential and commercial HVAC&R applications only during the last decade or so (Mehendale, 2013). In comparison with evaporators, it was thought that microchannel condenser coil technology was relatively easier to “transfer” from the automotive to the HVAC&R industry. This is because in normal operation, a condenser is expected to operate with no moisture condensing on the air side, whereas the air side surface of evaporators is routinely subject to moisture condensation. Additionally, the surface of evaporators used in heat pump applications is subject to frosting, which introduces additional complexities.

The standard round tube plate fin condenser coil has copper tubes mechanically bonded to aluminum fins with performance enhancements. More recently, aluminum tubes have also been used in round tube plate fin condenser designs to cut back on the high material costs associated with copper tubes. In contrast to round tube plate fin coils, microchannel coils are constructed utilizing an all-aluminum brazed fin construction. In microchannel coils, the tubes are flat instead of being round and one large port is replaced by several smaller ports (see Figure 1(a), which shows the general heat exchanger construction and 1(b) in which key geometrical features of the tube and fin are depicted). In this work, the thermal and hydrodynamic aspects of the air and refrigerant flows have been mathematically modeled for a configuration similar to the one depicted in Figure 1.

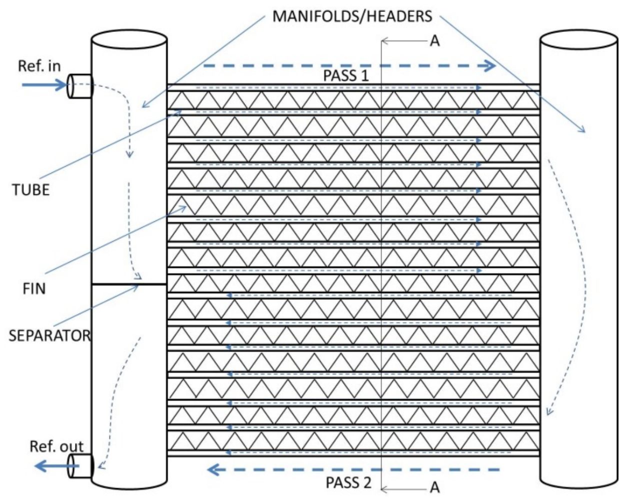

Figure 1(a): Basic building blocks of microchannel condenser

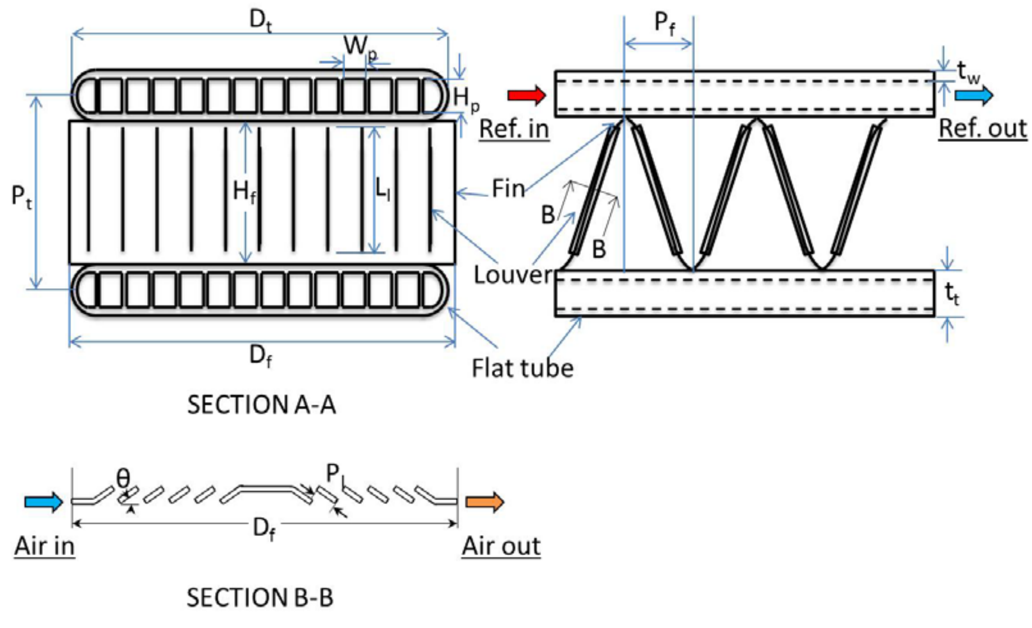

Figure 1(b): Key dimensions of microchannel condenser

As shown in Figures 1(a) and (b), a typical single row microchannel condenser coil is composed of three basic components:

1. A flat microchannel tube containing small ports through which refrigerant flows.
2. Corrugated louvered fins sandwiched between the microchannel tubes, through which air flows, and
3. Two refrigerant manifolds for collecting and distributing refrigerant to the tubes.

It is evident from Figure 1 that the air and refrigerant flow through the microchannel condenser in a cross-flow arrangement. These components are joined using an aluminum-zinc alloy brazing material in a nitrogen-charged braze furnace to produce the completed microchannel coil. Different coil circuiting patterns or pass arrangements are accomplished by appropriately placing and brazing separator discs in the distribution manifolds to feed the refrigerant through selected groups of flat tubes. The flat tubes allow maximization of the airside heat transfer surface area, and the multiple tiny refrigerant channels within the flat tubes maximize refrigerant side heat transfer by providing increased primary surface area and enhanced heat transfer coefficients. The metallurgical fin-tube bond resulting from the braze operation is designed to further enhance the thermal contact between the tubes and the fins. Due to all these constructional features, the thermal performance of the microchannel coil is greatly enhanced. However, in order to achieve the best possible heat duty, it is of the utmost importance to select the right refrigerant circuiting or pass arrangement.

Compared to traditionally applied round tube-plate fin heat exchangers, some of the benefits of microchannel heat exchangers include:

1. Improved or comparable heat transfer and about 65% lower refrigerant pressure drop. (Park and Hrnjak, 2008)
2. Up to about 10% lower refrigerant charge, due to the smaller internal volume of microchannel coils. (Park and Hrnjak, 2008)
3. Compactness of design, i.e., about two to three times higher surface area-to-volume ratio, (Garimella, 2003) providing opportunities for weight reduction and hence, cost reduction.

Today, simulation software that can accurately predict the performance of microchannel condensers is a very appropriate tool for designing such products in which complex, two-phase flows occur. These tools enable engineers and product designers to save significant time and expenses, by minimizing the use of expensive test benches. Currently, several models or simulation tools for heat exchangers are available in the literature. Yin et al. (2001) developed a finite volume (segment-by-segment) first principles-based CO_2 gas cooler model. They employed empirical correlations to predict the heat transfer coefficients, pressure drop, and fin efficiency. Jiang et al. (2002) presented a simulation and optimization tool for the design of air-cooled microchannel heat exchangers. The tool adopted an effectiveness-NTU method to simulate the dry surface condition and used the enthalpy potential method for simulating wet surface conditions. The tool, CoilDesigner^{TM}, version 3.9.20141.203 (Jiang et al., 2006), incorporates a network strategy for conveniently designing and analyzing coil circuiting. A segment-by-segment approach has been implemented within each tube to account for two-dimensionally non-uniform air flow distribution across the coil face. The model captures the significant change of refrigerant properties between vapor, two-phase, and liquid regimes. The software also provides a user-friendly graphical interface, and offers the user the choice of a wide variety of working fluids and heat transfer and pressure drop correlations.

In most early modeling works, the ports are assumed to have identical heat transfer processes and uniform refrigerant mass flow and pressure drop. The control volume (segment-by-segment) has been refined from a tube to a single port in two of the recent publications (Shao et al., 2009; Ren et al., 2013). In addition to the refined segment, the heat conduction effect has been considered in several models (Asinari et al., 2004; Shao et al., 2009; Martinez-Ballester et al., 2013; Ren et al., 2013). These models investigate tube-to-tube conduction, heat conduction between ports, as well as heat conduction in the refrigerant flow direction.

Schwentker et al. (2005) verified the prediction of CoilDesigner^{TM} against experimentally measured data for eight microchannel condensers with R-134a as the working fluid. The model was able to predict the condenser heat load within 2.25% for 80% of the 35 experimental data points. The average error, average absolute error, and the maximum error in the heat load prediction were -0.84%, 1.6%, and 4.6%, respectively.

More recently, in one of the most comprehensive microchannel condenser and gas cooler performance validation efforts, Huang et al. (2014) validated CoilDesigner^{TM} against 227 experimental data points for eight different working fluids including R410A and eighteen microchannel heat exchanger geometries from seven different sources of data. The average absolute deviation between the predicted and measured values of the heat duty and the refrigerant pressure drop was found to be 2.7% and 28%, respectively.

Very few studies have been reported in the open literature to understand the principles behind designing the optimal refrigerant circuiting in single row microchannel condensers. Huen and Dunn (1996) examined the effects of microchannel port diameter and shape on refrigerant circuit design in microchannel condensers. They conducted a single-phase heat exchanger analysis, which led them to conclude that smaller port sizes result in reduced heat exchanger internal volume and necessitate additional parallel refrigerant passages with reduced tube lengths. Port shape was shown to have a significant impact on heat exchanger volume and refrigerant circuiting. It was also demonstrated that the single-phase analysis was applicable to two-phase situations provided convective effects (e.g., annular flow condensation and convective boiling) were dominant. In a companion paper, Huen and Dunn (1996) extended their investigation to include the effect of refrigerant pressure drop and a cross-flow arrangement on the microchannel heat exchanger performance. They found that for a given port diameter, the pressure drop variation caused an optimum relationship between the number of parallel refrigerant passages and the heat exchanger length. Their analysis also led to the conclusion that an optimum combination of the number of ports and the number of tubes exists that minimizes the condenser volume for a given port diameter.

Ye et al. (2009) described and analyzed a novel design of multiple parallel-pass (MPP) microchannel tube condenser and its applications to automotive A/C systems. They introduced a flow distributor in the MPP condenser to enable parallel flow arrangement in adjacent flow paths. Through the MPP design, the two-phase zone was effectively enlarged to enhance the condensation heat transfer and reduce pressure drop. Performance test results showed that the MPP condenser had a heat duty up to 9.5% higher than the traditional benchmark microchannel condenser (also known as a parallel-flow condenser).

Thus, based on a review of the published literature, it is seen that the trends and principles governing the selection of the appropriate pass or circuiting arrangement for microchannel condensers have not been explored in depth. The influence of the microchannel condenser tube length on the optimal pass arrangement has also not been addressed. For instance, a question commonly facing thermal system designers is the following: for a given microchannel condenser tube and louvered fin design, a fixed number of tubes, and pre-defined air velocity and refrigerant flows and conditions, should a two-pass, a three-pass, or a four-pass circuit arrangement be preferred? Again, for any such arrangement, how should the microchannel tubes be proportioned among the various passes? Normally, an extensive experimental effort combined with simulation would be necessary to answer this question satisfactorily, and this necessarily involves significant expenditure of time and resources. In this article, we have applied the capabilities of CoilDesigner to explore the answers to these questions.

2. PASS ARRANGEMENT ANALYSIS

In this section, the influence of microchannel condenser tube length and pass arrangement on their thermal-hydraulic performance has been investigated. The microchannel tube and louvered fin geometrical parameters and the air side and R-410A side operating conditions selected for this study have been summarized in Table 1. R-410A was selected for the analysis because it is one of the most commonly used refrigerants today in residential and commercial HVAC&R heat exchangers. Table 2 lists the airside and refrigerant side pressure drop and heat transfer correlations used in this analysis.

Table 1: Microchannel condenser tube and fin geometry and air / refrigerant operating conditions used to analyze pass arrangements

Tube Length (m)	1.50 – 2.25
Total number of tubes	64
Tube depth, D_t (m)	0.0255
Tube thickness, t_t (m)	0.0021
Port Height x Port Width, H_p x W_p (m x m)	0.0015 x 0.0015
Number of ports per tube	14
Number of passes	2, 3, and 4
Fin density, (fins per inch)	20
Louver length, L_l (m)	0.00555
Louver angle, θ (deg.)	30
Louver pitch, P_l (m)	0.0010
Fin height (m)	0.00590
Air face velocity (m/s)	2.0 m/s
Inlet air temperature (°C)	35
R-410A inlet temperature (°C)	45
R-410A inlet superheat (°C)	20
R-410A mass flow rate (kg/s)	0.1000

Table 2: Heat transfer and pressure drop correlations used with CoilDesigner pass arrangement analysis

	Air side	Refrigerant side		
		Vapor	Two-Phase	Liquid
Heat transfer correlation	Chang and Wang (1997)	Gnielinski (1976)	Shah (1979)	Gnielinski (1976)
Pressure drop correlation	Chang et al. (2000)	Churchill (1977)	Friedel (1979)	Churchill (1977)

The microchannel condenser design selected for simulation has a configuration and dimensions similar to heat exchangers typically considered for application in residential and commercial air-conditioning applications. The selection of port dimensions is typically the result of tradeoffs among the desired pressure drop, heat transfer per circuit, number of circuits and refrigerant charge. Such a realistic configuration and dimensions were chosen for simulation so that the refrigerant circuit optimization results presented later on in this section would be practical and useful to researchers in the field as well as practical designers of microchannel condensers. It should be noted that over the range of condenser tube lengths explored in this analysis, the total heat transfer of a one-pass heat exchanger is always less than that of any two-, three-, or four-pass configuration. This is why only two, three, and four-pass circuit configurations have been considered in the following discussion. The following assumptions have been made in the analysis:

1. Air flow has been assumed to be uniform across the coil face in this study. This is because air side face velocity non-uniformities in practical applications arise from condenser and fan configuration, duct design, and other similar factors which are beyond the scope of this work.
2. Refrigerant mass flow and vapor quality maldistribution among the tubes of the microchannel passes, as well as port-to-port maldistribution, induced by header pressure drop are also ignored. In actual microchannel heat exchanger designs, header pressure drop will be influenced by header design parameters, the most important of which are header diameter and length, and is subject to turning losses, inlet/exit losses, and the offsetting effects of friction and deceleration, as refrigerant enters each successive tube in the circuit and the mass flux in the remainder of the header decreases. Refrigerant headers can be designed such that flow maldistribution could be minimized in condensers having exterior package dimensions typical of those in use today. In the present study, it is assumed that the refrigerant mass flux in the headers is neither too low (as this would tend to cause gravity-induced liquid–vapor stratification), nor too high (since this would cause variable pressure drop across the tubes of a given pass). Thus, both these extremes would lead to non-uniform refrigerant mass flow and vapor quality distribution among the tubes of a pass, and are avoided in this analysis.

However, it is recognized that flow and vapor quality maldistribution might be important for certain microchannel condenser design configurations. It is proposed to address the effects of refrigerant maldistribution on the optimum pass arrangements in a future article.

3. RESULTS AND DISCUSSION

A large number of two-, three-, and four-pass circuit configurations were analyzed in this work. For a given number of passes in the microchannel condenser, two basic types of pass configurations were investigated:
1. A contracting pass arrangement, and
2. An expanding pass arrangement.

For clarifying what is meant by a contracting pass arrangement, consider a p-pass arrangement with N[i] tubes in pass i, where i varies from 1 for the inlet or first pass to p for the outlet or p^{th} pass. A contracting pass arrangement is any pass arrangement that satisfies the following condition per equation (1):

$$N[1] \geq N[2] \geq N[3] \ldots \ldots . . \geq N[p] \quad (1)$$

Similarly, an expanding pass arrangement is any pass arrangement that satisfies the condition

$$N[1] \leq N[2] \leq N[3] \ldots \ldots .. N[p] \quad (2)$$

All possible contracting and expanding pass arrangements were automatically generated for the two-, three-, and four-pass circuit arrangements using a self-developed code written for this purpose. 52 two-pass, 398 three-pass, and 1,064 four combinations were simulated using the parametric pass arrangement analysis capability of CoilDesigner. It should be noticed that for microchannel condensers, a contracting pass arrangement is typically employed, while an expanding pass arrangement is most commonly built into industrial applications of microchannel evaporators. The reason for this practice is that in condensers, the density of the refrigerant progressively increases as it flows through the condenser, thus necessitating successively smaller passes in the refrigerant flow direction. However, with a tool such as CoilDesigner at our disposal, it is easier to select the correct pass arrangements that lead to optimal heat duty, whether this pass arrangement is an expanding or a contracting one, as will be shown later in this section.

Figures 2, 3, 4, and 5 show the heat duty (in kW) plotted as a function of the refrigerant pressure drop (in kPa) for the two-, three-, and four-pass contracting and expanding pass arrangement configurations. These figures represent coil thermal performance data for tube lengths of 1.50 m, 1.75 m, 2.00 m, and 2.25 m, respectively. By plotting the data in the format depicted in the figures, several important trends can be inferred. First, within a given pass arrangement configuration, it is clear that, in general, designs that result in a lower pressure drop lead to a higher heat duty. This observation is consistent with the fact that as the refrigerant pressure drop through the condenser decreases, the average temperature difference across the cross-flow heat exchanger between the refrigerant and the cooling air increases. Since it is this temperature difference that drives the heat transfer, the trends apparent from Figures 2 through 5 are reasonable. The second observation that can be made from the figures is that the 2-, 3-, and 4-pass expanding circuit arrangements span a much wider range of refrigerant pressure drops, and hence, heat duties, in comparison with the contracting pass arrangements. This implies that the expanding pass configurations display a greater sensitivity to thermal performance changes than the contracting pass configurations. From a practical viewpoint, the most important fact that is obvious from Figures 2 through 5 is that, compared to the expanding pass configurations, the contracting pass configurations with the same number of passes result in higher heat duty for comparable refrigerant pressure drop, with the exception of the two-pass configurations in Figures 2 and 3. It should be noted that this observation is always true whenever the refrigerant exits the condenser as subcooled liquid, which is the case for Figures 4 (except for the two worst performing pass arrangements) and 5 (except for the worst performing pass arrangement). In Figures 2 and 3, the heat duty for all pass configurations is such that two-phase refrigerant exits the condenser. In Figure 2, the outlet vapor quality ranges from about 16% to 30%, while in Figure 3, the same varies from about 1% for the best performing pass configuration to about 20% for the pass arrangement with the lowest heat transfer.

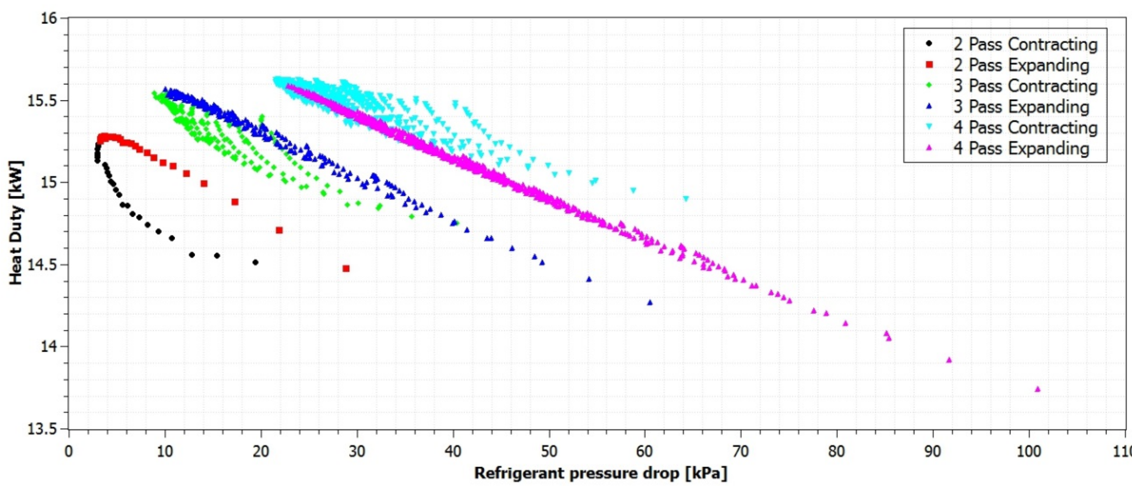

Figure 2: Thermal performance characteristics of microchannel condenser with a 1.50 m tube length

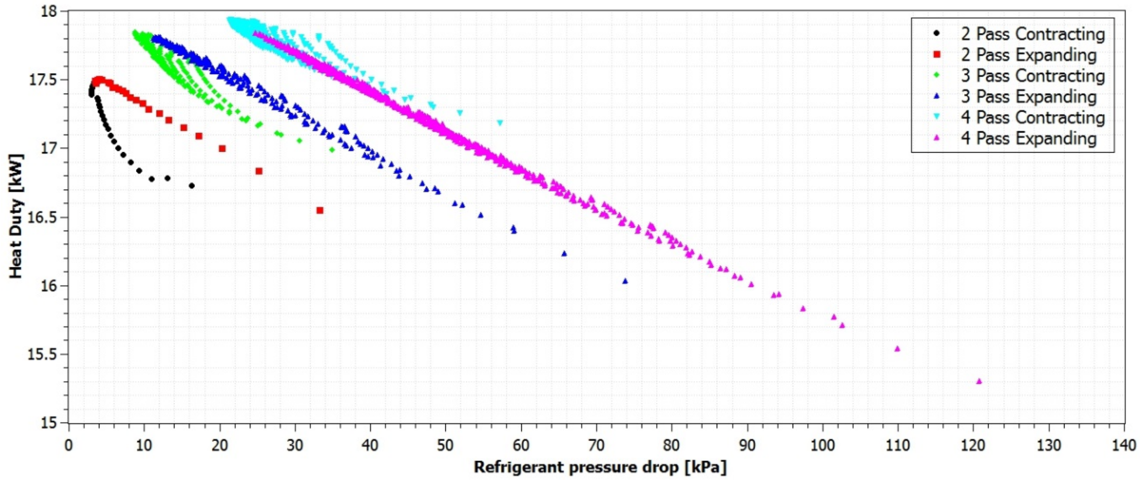

Figure 3: Thermal performance characteristics of microchannel condenser with a 1.75 m tube length

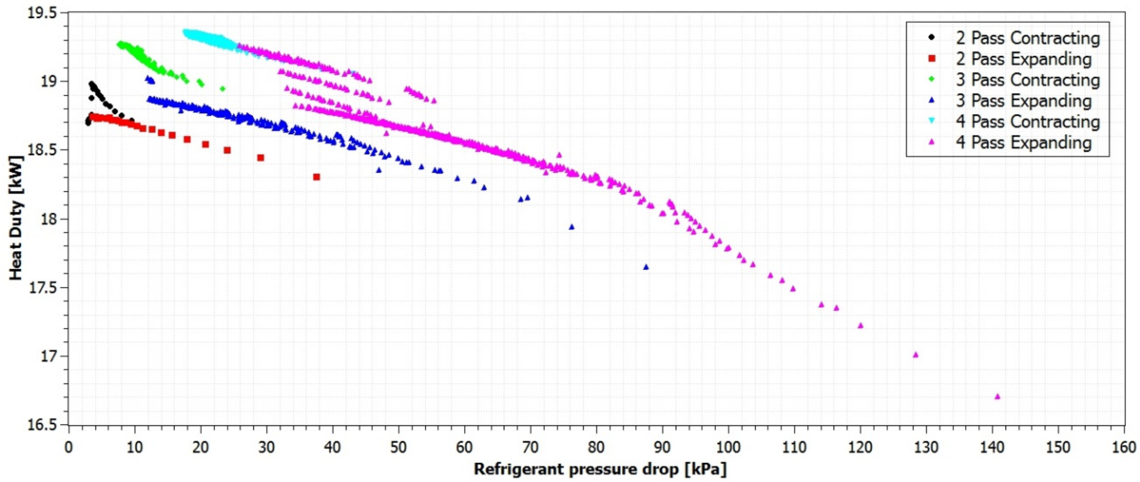

Figure 4: Thermal performance characteristics of microchannel condenser with a 2.00 m tube length

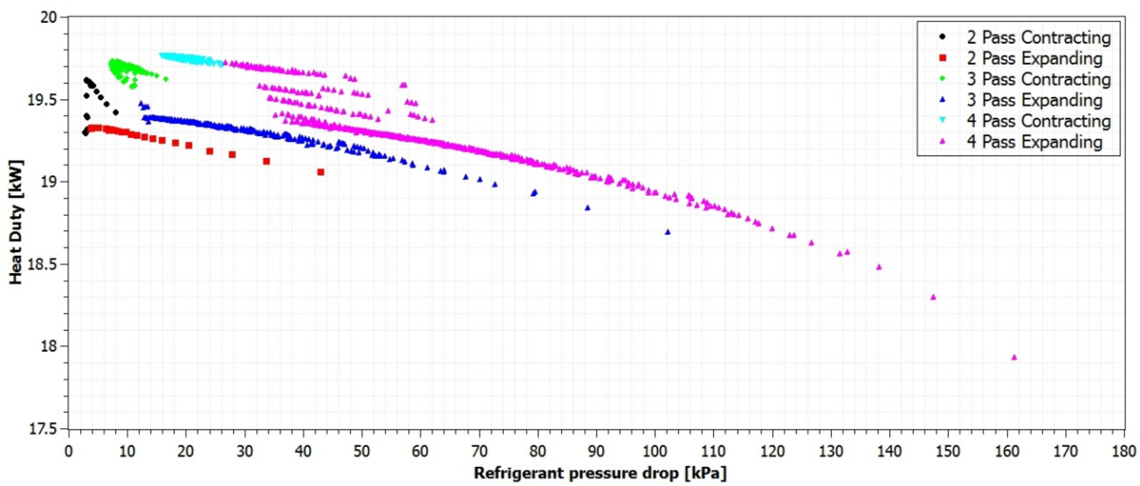

Figure 5: Thermal performance characteristics of microchannel condenser with a 2.25 m tube length

For a 2.00 m tube length condenser, Columns 3, 4, and 5 of Table 3 show the best performing pass arrangements and their heat duty and refrigerant pressure drop in each of the six classes of pass configurations, i.e., 2-pass contracting and expanding, 3-pass contracting and expanding, and 4-pass contracting and expanding. The results for the 2.25 m long condenser are similar to those presented in this table. Columns 6, 7, and 8 represent the lowest performing configurations in each class of circuit configurations. As seen from Table 3, the 38%/31%/20%/11%, 42%/41%/17%, 23%/25%/25%/27%, 32%/34%/34%, 67%/33%, and 47%/53% configurations are the best performing pass arrangements, in that order. Between the best and the worst performing of these top six pass arrangements, there is a difference in heat duty of more than 3%. As seen from Table 3, a more or less even distribution of tubes among the four passes of a 4-pass expanding arrangement tends to maximize its heat duty. If the pass arrangement is made very uneven, by making the last pass the biggest (for example, 9/9/9/73), there will be a significant (about 13%) penalty in heat duty, due to the non-optimal utilization of heat transfer surface area, and the excessive vapor refrigerant pressure drop in the first and second passes. However, from an application perspective, the pressure drop for the 4-pass expanding configuration may be excessively large, and hence, this arrangement is not preferred. The 3-pass contracting pass arrangement may also be used with minimal loss of heat duty compared to the contracting 4-pass optimum circuit arrangement. However, in real applications, considerations of inlet and outlet plumbing being on the same side of the coil often dictate the choice of pass arrangement. In such cases, the 4-pass contracting arrangement will be the circuit arrangement of choice. As is apparent from Table 3, the 2-pass contracting and expanding configurations will lead to up to 3% loss in heat duty, and hence, should not be used. Overall, the four pass contracting configuration, with its optimal heat duty-refrigerant pressure drop characteristics will be the preferred pass arrangement.

Table 3: Best and worst performing pass arrangements for a condenser with 2.0 m tube length

Tube length (m)	Pass configuration	Pass arrangement (% tubes) for maximum heat duty	Maximum heat duty [W]	Corresponding refrigerant ΔP (kPa)	Pass arrangement for minimum heat duty	Minimum heat duty [W]	Corresponding refrigerant ΔP (kPa)
2.0	4 contracting	38/31/20/11	19358	17.7	73/9/9/9	19057	43.1
	3 contracting	42/41/17	19274	7.8	80/10/10	18944	23.3
	4 expanding	23/25/25/27	19264	25.8	9/9/9/73	16709	140.7
	3 expanding	32/34/34	19027	11.9	10/10/80	17649	87.5
	2 contracting	67/33	18981	3.5	63/37	18700	2.9
	2 expanding	47/53	18748	3.7	9/91	18309	37.5

4. CONCLUSIONS AND FUTURE WORK

The trends and principles governing the selection of the appropriate pass or circuiting arrangement for microchannel condensers have been explored in this article. For a given microchannel condenser tube and louvered fin design, a fixed number of tubes, and fixed air and refrigerant inlet conditions, two-pass, three-pass, and four-pass circuit arrangements with contracting and expanding pass designs were generated and simulated using the parametric pass arrangement analysis capability built into CoilDesigner. Air velocity maldistribution on the coil face and refrigerant mass flow and vapor quality maldistribution among the tubes of the microchannel passes, as well as port-to-port maldistribution, induced by header pressure drop have been ignored.

The simulations were conducted for tube lengths of 1.50 m, 1.75 m, 2.00 m, and 2.25 m. It was clearly seen that, in general, among the designs of a particular pass configuration, designs that result in a lower total coil pressure drop lead to a higher heat duty. This is due to the fact that as the refrigerant pressure drop through the condenser decreases, the average temperature difference between the refrigerant and the cooling air that drives the heat transfer also increases. The two-pass, three-pass, and four-pass expanding pass arrangements were seen to span a much wider range of refrigerant pressure drops, and hence, heat duties, compared to the respective contracting pass arrangements with the same number of passes. Thus, the expanding pass configurations are more sensitive to thermal performance changes than the contracting pass configurations. Again, compared to the expanding pass configurations, the contracting pass configurations with identical number of passes yield higher heat duty for comparable refrigerant pressure drop. This finding was always true whenever the refrigerant leaving the condenser was in the subcooled liquid phase.

Guidelines have also been provided to aid researchers and thermal system designers select the optimal pass arrangement for microchannel condensers similar to the one considered in this study. The 38%/31%/20%/11%, 42%/41%/17%, 23%/25%/25%/27%, 32%/34%/34%, 67%/33%, and 47%/53% configurations were the best performing pass arrangements, in that order. Between the 38%/31%/20%/11% and the 47%/53% pass arrangements, the heat duty differs by more than 3%. If the 4-pass expanding pass arrangement is made very uneven, by making the last pass the biggest, there will be a significant (about 13%) penalty in heat duty compared to the best pass arrangement in the same class, due to the non-optimal utilization of heat transfer surface area, and the excessive vapor refrigerant pressure drop in the first and second passes. A more or less even distribution of tubes among the four passes of a 4-pass expanding arrangement tends to maximize its heat duty. However, the pressure drop for the 4-pass expanding configuration may be excessively large for practical applications, and hence, this arrangement is not preferred. The three pass contracting pass arrangement may also be used with minimal loss of heat duty compared to the contracting 4-pass optimum circuit arrangement. However, in real applications, considerations of inlet and outlet plumbing often dictate the choice of the pass arrangement. In such cases, the four pass contracting arrangement will be the circuit arrangement of choice, since, in addition to its superior heat duty and moderate refrigerant pressure drop, its inlet and outlet pipes will be on the same side of the coil, a configuration preferred for its convenience and compact packaging. As is apparent from Table 3, the 2-pass contracting and expanding configurations will lead to up to 3% loss in heat duty, and hence, should not be used. Overall, the 38%/31%/20%/11%, configuration, with its optimal heat duty-refrigerant pressure drop characteristics will be the preferred pass arrangement.

In future development of this work, we plan to explore the influence of different microchannel tube geometries, a wider range of coil dimensions, other air and refrigerant flow rates and conditions, and air side and refrigerant side flow maldistribution on the optimal circuit arrangements.

NOMENCLATURE

D_f	fin depth	(m)
D_t	tube depth	(m)
H_f	fin height	(m)
H_p	port (channel) width	(m)
i	pass number beginning with inlet pass	(-)
L_l	louver length	(m)
N[i]	number of tubes in pass i	(-)
P	maximum number of passes	(-)
P_f	fin pitch	(m)
P_l	louver pitch	(m)
P_t	tube pitch	(m)
t_f	fin thickness	(m)
t_t	tube thickness	(m)
t_w	tube wall thickness	(m)
θ	louver angle	(deg.)
W_p	port (channel) width	(m)

REFERENCES

Asinari, P., Cecchinato, L., Fornasieri, E., 2004, Effect of Thermal Conduction in Microchannel Gas Coolers for Carbon Dioxide, *Int. J. Refrig.*, vol. 27: p. 577-586.

Chang, Y., Chang, H., Tsai, L., Wang, C., 2000, A Generalized Friction Correlation for Louver Fin Geometry, *Int. J. Heat Mass Tran.*, vol. 43, no. 12: p. 2237-2243.

Chang, Y., Wang, C., 1997, A Generalized Heat Transfer Correlation for Louver Fin Geometry, *Int. J. Heat Mass Tran.*, vol. 40, no. 3: p. 533-544.

Churchill, S., 1977, Friction-Factor Equation Spans All Fluid-Flow Regimes, *Chem. Eng*., vol. 84: p. 91-92.
Garimella, S., 2003, Innovations in Energy Efficient and Environmentally Friendly Space-Conditioning Systems, *Energy*, vol. 28: p. 1593–1614.
Gnielinski, V., 1976, New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, *Int. Chem. Eng.*, vol. 16, no. 2: p. 359-368.
Huang, L., Aute, V., Radermacher, R., 2014, A Model for Air-To-Refrigerant Microchannel Condensers with Variable Tube and Fin Geometries, *Int. J. Refrig*., vol. 40: p. 269-281.
Huen, M., Dunn, W., 1996, Principles of Refrigerant Circuiting with Application to Microchannel Condensers: Part I - Problem Formulation and the Effects of Port Diameter and Port Shape, *ASHRAE Trans.*, ASHRAE, Atlanta, GA, vol.102: p. 373-381.
Huen, M., Dunn, W., 1996, Principles of Refrigerant Circuiting with Application to Microchannel Condensers: Part II - The Pressure-Drop Effect and the Cross-Flow Heat Exchanger Effect, *ASHRAE Trans*., ASHRAE, Atlanta, GA, Vol.102: p. 382-383.
Jiang, H., Aute, V., Radermacher, R., 2002, A User-Friendly Simulation and Optimization Tool for Design of Coils, *Proc. 9th Int. Refrig. Air Cond. Conf. at Purdue*, Purdue University, West Lafayette, IN, Paper R5-1.
Jiang H., Aute V., Radermacher R., 2006, CoilDesigner: A General Purpose Simulation and Design Tool for Air to Refrigerant Heat Exchangers, *Int. J. Refrig.*, vol. 29, no. 4: p. 601-610.
Martinez-Ballester, S., Corberan, J., Gonzalvez-Macia, J., 2013, Numerical Model for Micro-Channel Condensers and Gas Coolers: Part I - Model Description and Validation, *Int. J. Refrig.*, vol. 36: p. 173-190.
Mehendale, S., 2013, Principles of Refrigerant Circuiting in Single Row Microchannel Evaporators, *Proc. ASME 11th Int. Conf. Nanochannels, Microchannels, and Minichannels (ICNMM 2013, Sapporo, Japan)*, ASME, New York, NY, Paper No. ICNMM2013-73137, doi: 10.1115/ICNMM2013-73137.
Park, C., Hrnjak, P., 2008, Experimental and Numerical Study on Microchannel and Round-Tube Condensers in a R410A Residential Air-Conditioning System, *Int. J. Refrig.*, vol. 31: p. 822-831.
Ren, T., Ding, D., Wang, T., Hu, H., 2013, A General Three Dimensional Simulation Approach for Micro-Channel Heat Exchanger Based on Graph Theory, *Appl.Therm.Eng.*, vol. 59: p. 660-674.
Schwentker, R.A., Aute, V.C., Radermacher, R., Mercer, K.B., 2005, Simulation and Design Tool for Microchannel Heat Exchangers, *Proc. Fifth Int. Conf. Enhanced, Compact, and Ultra-Compact Heat Exchangers: Science and Technology*, Eds. R.K. Shah, M. Ishizuka, T.M. Rudy, and V.V. Wadekar, Hoboken, NJ, USA., Paper CHE 2005-24.
Shah, M., 1979, A General Correlation for Heat Transfer during Film Condensation inside Pipes, *Int. J. Heat Mass Tran.*, vol. 22: p. 547-556.
Shao, L., Yang, L., Zhang, C., Gu, B., 2009, Numerical Modeling of Serpentine Microchannel Condensers, *Int. J. Refrig.*, vol. 32: p. 1162-1172.
Ye, L., Tong, M., Zeng, X., 2009, Design and analysis of multiple parallel-pass condensers, *Int. J. Refrig.*, vol. 32: p. 1153-1161.
Yin, J., Bullard, C., Hrnjak, P., 2001, R-744 Gas Cooler Model Development and Validation, *Int. J. Refrig.*, vol. 24: p. 692-701.